Calhoun: The NPS Institutional Archive
DSpace Repository

Theses and Dissertations 1. Thesis and Dissertation Collection, all items

1950

A study of pressure control for a blowdown wind tunnel

Kanaga, F. N.; Rice, D. B.

University of Michigan

http://hdl.handle.net/10945/24862

Downloaded from NPS Archive: Calhoun

Calhoun is the Naval Postgraduate School's public access digital repository for research materials and institutional publications created by the NPS community. Calhoun is named for Professor of Mathematics Guy K. Calhoun, NPS's first appointed -- and published -- scholarly author.

Dudley Knox Library / Naval Postgraduate School
411 Dyer Road / 1 University Circle
Monterey, California USA 93943

http://www.nps.edu/library

Thesis
K13

Library
U. S. Naval Postgraduate School
Annapolis, Md.

A STUDY OF PRESSURE CONTROL FOR A BLOWDOWN WIND TUNNEL

A
THESIS
SUBMITTED TO
THE DEPARTMENT OF AERONAUTICAL ENGINEERING
FOR
MASTER OF SCIENCE DEGREE IN ENGINEERING
BY
F. N. KAHALA AND C. E. RICE

12984

UNIVERSITY OF MICHIGAN
JULY 1950

Thesis
K13

A STUDY OF PRESSURE CONTROL FOR A BLOWDOWN WIND TUNNEL

SUMMARY

The problem involved in this investigation was to determine the characteristics of a system that would effectively control the settling chamber pressure within certain arbitrary limits.

The study was concerned only with one particular installation. However, the methods employed should be applicable to any similar problem.

As a result of this investigation it may be concluded that pressure control to within one pound per square inch may be obtained within approximately one second. This assumes the use of a servomechanism in which the control valve is moved by means of a servomotor acting through a gear train and rack. The general features of the servomechanism are listed in Table II. A motor of such characteristics as are required to provide this control is described in this report.

It is recommended that such a servomechanism be designed and installed in the subject wind tunnel

in the place of the present installation. It is expected that this will be accomplished in the near future.

This investigation was conducted in the laboratories of the Aeronautical Engineering Department of the University of Michigan by F.N. Kanaga and D.B. Rice. It is submitted as a thesis for credit towards the degree of Master of Science.

INTRODUCTION

The problem to be investigated is the control of the air pressure in the settling chamber of a blowdown wind tunnel. The particular wind tunnel to be studied is already installed in the East Engineering Building of the University of Michigan at Ann Arbor, Michigan. A photograph of the installation is shown in Figure 1, and some of the physical measurements are in Table I.

The principal components of the wind tunnel are the storage tank, the driers, the quick-opening valve, the control valve, the settling chamber, the nozzle and diffuser, and the control system. Air pressure at about 95 pounds per square inch gage is furnished by the building air supply system.

The perfect control system in this case would be one that would measure the instantaneous air pressure in the settling chamber, compare it with the desired pressure, and then actuate the control valve so that the pressure would be changed to exactly the desired pressure in an arbitrarily short period of time. When the wind tunnel is employed for supersonic laboratory tests, the length of blowdown runs vary

from about 3.5 to about 25 seconds. This variation depends primarily upon the value of the desired pressure in the settling chamber, providing the initial supply tank pressure is constant. To make maximum use of this short time, it is desired that the pressure be stabilized within one second. From experience, it was decided that control of the settling tank pressure to within plus or minus one pound per square inch would be as close as would conceivably be required in the laboratory. Therefore, the arbitrary goal set for the control system is to reduce the pressure error (instantaneous settling chamber pressure minus desired pressure) to within plus or minus one pound per square inch within one second.

The present control system installation is of a type in general use in the chemical industry for temperature control and was modified for this pressure control application. This installation is not satisfactory for this wind tunnel for several reasons. The primary reason is that the pneumatic portion of the circuit is such that the control air pressures are too small to move the diaphragm actuated control valve with sufficient velocity to give close control of the

settling chamber pressure. With the control system on "automatic", it required 20.7 seconds for the valve to open fully, and 13.5 seconds for it to close. Increasing the control air pressures would involve very extensive reworking of the pneumatic system and does not appear to offer a substantial improvement due to the pressure limitations of the valve diaphragm.

In view of the above, another method of pressure control has been investigated and made the subject of this report. The employment of an electric motor to drive the valve stem through a gear train and rack was decided upon as a likely possibility for providing valve movement, this machinery to be the load for some sort of servomechanism. A sketch of this proposed arrangement is shown in Figure 2.

The method of investigation was to determine the forces that act on the valve end of the valve stem, and then to determine the control forces necessary to overcome these valve forces and to move the valve to a position corresponding to the desired pressure in a stable manner within the desired time limit.

The major tool employed in this study was the electronic differential analyser, (electronic

analog computer) which will be referred to hereafter in this report as the "computer". Information concerning this computer may be found in references (a), (b), (c), (d), and (e).

A differential equation was developed which embodies the forces that act on the valve. By making use of the various functions of the operational amplifiers, which constitute the computer, for sign changing, multiplication by a constant factor, integration and summation, the differential equation was duplicated on the computer. The voltages at various points within the computer represent the values of corresponding functions within the equation, and these voltages are fed through a Brush direct current amplifier to a Brush magnetic recording oscillograph, or recorder, so that the variation with time of the various functions in the equation was recorded on tape.

This investigation is limited to one specific wind tunnel. The methods used, however, should be applicable to other such systems.

This study was made during the 1949-1950 school year in the laboratories of the Aeronautical Engineering Department of the University of Michigan. The authors wish to acknowledge and express appreciation

for the invaluable guidance and suggestions given by Dr. L.L. Rauch of the Aeronautical Engineering Department.

PROCEDURE

WIND TUNNEL DATA

The variation of settling chamber pressure with valve position as shown in Plate I was found by presetting the control valve by manual means to various positions and then reading the maximum settling chamber pressure reached when the quick-opening valve was opened. These values were read from a servo-type electrical indicator which was actuated by a differential transformer attached to a bourdon tube located on the settling chamber. The initial tank pressure was constant for each set of runs.

The coulomb or dry friction and the weight of the moving parts of the valve were determined as indicated in Figure 3. The forces measured with the valve barely moving up were weight plus friction, and with downward motion they were weight minus friction. From this the weight was determined to be 10.38 pounds and the coulomb friction to be 1.23 pounds.

Lift forces on the valve due to air passing through the valve were measured by using apparatus set up as in Figure 4. A plot of lift forces versus valve position is on Plate II.

No attempt was made to measure the effect of viscous friction of the valve. The force necessary to start the valve moving very slowly was about one pound greater than the coulomb friction force. This force difference is sometimes called "sticktion".

COMPENSATING FOR THE LIFT FORCE

Referring to Plate II, it will be noted that the lift force due to the action of the air passing through the control valve has been approximated as a linearized function of valve position. The weight of the valve and stem will cancel out 10.38 pounds of this force. However, this still leaves a large force to be supplied by the control system when the valve is being operated near the fully opened position. By adding a six pound weight to the valve stem, the lift force is balanced by weight when the valve is one inch, or two-thirds of the way, open. This six pound weight is shown in Figure 2 as an addition to the weight of the valve stem. It will be noted that when the control valve is fully opened the uncompensated lift force will be six pounds.

SELECTION OF MOTOR

From Computation I, which is based in part on a rough assumption of accelerations, horsepower requirements for a motor to drive the control valve were found to be 0.000564 HP. A servomotor that would deliver about 10 times this horsepower at maximum output was tentatively selected. This was the Diehl, Navy type CDA 211052 motor for which the characteristic curves are shown on Plate III. If geared so that the motor would accelerate to 2000 rpm in 1/2 second, it was found that an average force of 20.5 pounds would be exerted on the valve stem, varying from 23.8 pounds at zero rpm to 15.3 pounds at 2000 rpm. It was estimated that the forces involved would be below these figures. The gear ratio necessary to obtain the average force is 69.7 to 1, that is, the gear that drives the rack should turn one revolution while the motor turns 69.7 revolutions.

After a solution to the problem was obtained, it was apparent that a motor which could exert a greater torque would be desireable in the interests of a larger margin of safety. The Diehl Navy type CDA 211426 or 211454, which was found to have about double the power output of the original motor, was chosen. Characteristic curves for this motor are shown on Plate III.

THE DIFFERENTIAL EQUATION

The forces on the valve stem were summed up in a differential equation as follows:

$$m\ddot{x} + C_1\frac{\dot{x}}{|\dot{x}|} + C_2\dot{x} + K_1\dot{p}_e + K_2 p_e + K_3\int_0^t p_e\,dt + K_4 P_d = 0 \qquad (1)$$

where:

x	is valve position in feet measured from closed to open. Full valve travel is 1.5 inches for this valve. For the coordinate system, see Figure 5.
m	is the mass in slugs of the moveable valve parts plus the six pound weight previously mentioned.
p	is the instantaneous pressure in pounds per square inch in the settling chamber.
P	is the desired pressure in pounds per square inch in the settling chamber.
p_e	is the pressure error in pounds per square inch in the settling chamber, or (p - P).
C_1	is the coulomb friction force in pounds.
$\frac{\dot{x}}{\lvert\dot{x}\rvert}$	is a unity term whose sign furnishes the direction of the coulomb friction force.
C_2	is a coefficient of viscous friction, with units of pound seconds/feet.

K_1, K_2 and K_3 are constants which when multiplied by $\dot{p}_e$, p_e and $\int_0^t p_e dt$ respectively represent the forces in pounds to be supplied to the control valve stem by the control mechanism.

$K_4 P_d$ is a disturbing force in pounds on the valve itself.

In order to reduce equation (1) to a function of one variable, the relation obtained from Plate I is used:

$$p = n_1 + n_2 x \tag{2}$$

where n_1 is the ordinate at $x = 0$ and n_2 is the slope of the line. Referring to Plate I, it is seen that equation (2) is valid only for the portion of the curve below a valve opening of about 0.875 inches. The normal operating region for the control valve is in the straight line portion of the curve, therefore the linear relationship between settling chamber pressure and valve opening was used. Thus from the above:

$$p_e = p - P \tag{3}$$

$$\dot{p}_e = \dot{p} = n_2 \dot{x}, \qquad \dot{x} = \dot{p}_e / n_2 \tag{4}$$

$$\ddot{p}_e = \ddot{p} = n_2 \ddot{x}, \qquad \ddot{x} = \ddot{p}_e / n_2 \tag{5}$$

Substitution of (4) and (5) into (1) gives a differential equation with p_e/n_2 as the variable.

$$m\left(\frac{\ddot{p}_e}{n_2}\right) + C_1\frac{\dot{p}_e}{|\dot{p}_e|} + C_2\left(\frac{\dot{p}_e}{n_2}\right) + K_1 n_2\left(\frac{\dot{p}_e}{n_2}\right) + K_2 n_2\left(\frac{p_e}{n_2}\right) + K_3 n_2\int_0^t\left(\frac{p_e}{n_2}\right)dt + K_4 P_d = 0 \tag{6}$$

By defining $K_1 n_2$ as k_1, $K_2 n_2$ as k_2 and $K_3 n_2$ as k_3, equation (6) becomes more simple in form:

$$m\left(\frac{\ddot{p}_e}{n_2}\right) + C_1\frac{\dot{p}_e}{|\dot{p}_e|} + C_2\left(\frac{\dot{p}_e}{n_2}\right) + k_1\left(\frac{\dot{p}_e}{n_2}\right) + k_2\left(\frac{p_e}{n_2}\right) + k_3\int_0^t\left(\frac{p_e}{n_2}\right)dt + K_4 P_d = 0 \tag{7}$$

The various terms of the differential equation represent physical effects as follows:

$m\left(\frac{\ddot{p}_e}{n_2}\right)$	is the effect of the mass of the moving parts of the control valve plus the six pound compensating weight.
$C_1\frac{\dot{p}_e}{\lvert\dot{p}_e\rvert}$	is the coulomb friction force, of magnitude C_1 and varying in sign according to the sign of $\dot{p}_e$.
$C_2\left(\frac{\dot{p}_e}{n_2}\right)$	is the viscous friction force caused by the motion of the control valve.
$k_1\left(\frac{\dot{p}_e}{n_2}\right)$	is the derivative term or the error-rate viscous damping force to be supplied to the valve stem by the controller.
$k_2\left(\frac{p_e}{n_2}\right)$	is the direct proportional force to be supplied by the controller.
$k_3\int_0^t\left(\frac{p_e}{n_2}\right)dt$	is the integral control force to be supplied by the controller.

K_4P_d is a disturbing force applied to the valve stem itself, and may represent the unbalanced lift force on the valve which was previously discussed in this section.

THE TWO CASES OF THE DIFFERENTIAL EQUATION

Equation (7) is the general equation which represents the system of forces which are summed at the valve stem. Two solutions to the equation were obtained, Case I, in which the K_4P_d term and the tank pressure during the run were kept constant, and Case II, in which the K_4P_d term varied with the valve position as shown in Plate II, and the tank pressure decreased with time during the run. While Case I does not accurately describe the forces acting on the control valve, it is simple, and its solution gave the authors experience in operating the relatively complicated computer.

The actual computer circuit for Case I is shown on Plate IV. It will be noted that the invariant K_4P_d term is simulated by a constant voltage fed to the summing point through a one megohm resistor. The initial conditions for the equation were set into the apparatus as follows: prior to zero time, switches A, B and C were closed, so that the condensers across amplifiers B and C were shorted, and could have no initial charge.

It is evident that this condition must prevail since both velocity and integral control must be zero at zero time. The condenser across amplifier 4 was initially charged up to the voltage of battery D, which has the effect of applying an initial value of $\frac{p_e}{n_2}$ at time equal to zero. Runs were started by closing the three switches simultaneously.

It was assumed that in the operation of the wind tunnel the control system will be in operation before the quick-opening valve is opened. Thus when a desired pressure input is introduced into the controller with the quick-opening valve closed, the controller will open the valve to full open position in an effort to increase the settling chamber pressure, which of course will remain at zero pounds per square inch gage. It will be necessary to keep the integral control from operating prior to the opening of the quick-opening valve. Failure to do this would build up a large integral control force. When the quick-opening valve is opened, the settling chamber pressure reaches its maximum value in a short time. In this problem, it is assumed that the time required for the opening of the quick-opening valve and for the settling chamber pressure to rise to its maximum value

is zero. This assumption imposes a more severe condition on the controller than actually exists. The actual problem, however, is simplified through use of this assumption.

In Case I, as seen in Figures 7 through 12, the initial value of the voltage of the term $\left(\frac{p_e}{n_2}\right) \times 500$ is plus 30, which corresponds to an initial value of p_e of 35.8 psig when n_2 is 394 psig/ft for a supply tank pressure (p_t) of 93 psig. This value for initial pressure upset is about the maximum that would be encountered under actual operating conditions of the wind tunnel. This is accounted for by the fact that the settling chamber pressure must be above a minimum of 52 psig for a supersonic shock-free flow through the nozzle test section when the "diffuser" is in place.

On Plate V is the circuit diagram for the computer for Case II. Equation (7) is modified for this case to take into account the variation of K_4P_d with the control valve position. From Plate II,

$$K_4P_d = 11.88 - 142\,x \tag{8}$$

From equations (2) and (3),

$$x = \left(\frac{p_e}{n_2}\right) + \frac{P}{n_2} - \frac{n_1}{n_2} \tag{9}$$

Combining equations (8) and (9):

$$K_4 P_d = 11.88 - 142\left(\frac{p_e}{n_2}\right) - 142\,\frac{P}{n_2} + 142\,\frac{n_1}{n_2} \tag{10}$$

Thus equation (7) is modified for Case II as follows:

$$\begin{gathered} m\left(\frac{\ddot{p}_e}{n_2}\right) + C_1\,\frac{\dot{p}_e}{|\dot{p}_e|} + C_2\left(\frac{\dot{p}_e}{n_2}\right) + k_1\left(\frac{\dot{p}_e}{n_2}\right) + k_2\left(\frac{p_e}{n_2}\right) + \\ k_3\int_0^t\left(\frac{p_e}{n_2}\right)dt + 11.88 - 142\left(\frac{p_e}{n_2}\right) - 142\,\frac{P}{n_2} + 142\,\frac{n_1}{n_2} = 0 \end{gathered} \tag{11}$$

To simulate reduction of the tank pressure during a run, all the terms that contain the coefficient n_2 must be varied with time. On Plate VI are plots of n_1, n_2 and n_1/n_2 versus supply tank pressure which were derived from Plate I. On Plate VII are found plots of p_t/p_{t_i} for the supply tank versus time and σ versus time for two different σ's, with σ defined as the ratio of initial storage tank pressure to settling chamber pressure, or $(p_{t_i}/p)_{abs}$. It can be seen that [illegible]

equals 1.25, the length of the run is 3.5 seconds. This appears to be about as severe a case of pressure drop as would be encountered in actual operation of the wind tunnel. Assuming an initial tank pressure of 93 psig and a σ of 1.25, it follows that the desired settling chamber pressure is 71.5 psig, and the tank pressure at the end of the run would be about 75 psig.

A voltage to give an initial p_e of 21.5 psig, which corresponds to a desired settling chamber pressure of 71.5 psig was introduced into the computer circuit by means of battery D as indicated on Plate V. Switches A, B and C operate in the same manner as described for Case I.

Referring then to Plate VI, if the tank pressure will drop from 93 to 75 psig in 3.5 seconds, then n_2 must decrease from 594 to 512 psig/foot and n_1/n_2 must decrease from 0.0587 to 0.049 feet during the same time interval. This variation was assumed to be linear with time. From Plate VII it can be seen that for a small σ, the variation of p_t/p_{t_i} with time can be approximated by a straight line, so that the above assumption is nearly correct.

Referring to equations(6) and (11) and to

Plate V, it is noted that the term n_2 is contained as a direct proportionality factor in the terms k_1, k_2 and k_3 of the differential equation. Thus, by varying the voltage output of amplifier 6 in direct proportion to the value of n_2, the variations of k_1, k_2 and k_3 that were caused by the dropping of tank pressure were simulated.

Also in equation (11), the terms $142P/n_2$ and $142n_1/n_2$ were varied with time. In this case, P was taken as 71.5 psig as was previously derived. Then at zero time, the two terms were -17.1 and 8.3 volts repectively, and at zero plus 0.5 seconds, they were -19.9 and 7.0 volts.

Potentiometers were employed as in Plate V to vary simultaneously the voltages described in the above two paragraphs linearly with time. The three potentiometers were arranged in tandem on a single shaft, or "ganged", so that such control over the effect of tank pressure variation could be simulated.

In addition to appearing as a coefficient in several of the terms of the differential equation, the factor n_2 also appears as a part of the variable P_e/n_2 and its derivative and integral terms. No physical operation during the runs was employed to

vary the factor n_2 in this variable. Its linear variation with time was taken into account in evaluating the ordinates of the various functions of the variable from the recording tape.

RESULTS AND DISCUSSION

The time variation of the various pertinent terms of the differential equation, as obtained by use of the computer, is shown in Figures 7 through 12 for Case I, and in Figures 13 through 17 for Case II.

Values of k_1, k_2 and k_3 were chosen experimentally such that the pressure error decreased to a negligible amount in the shortest possible time without exceeding the allowable forces that the motor is capable of transmitting to the valve stem. The values of the resistances $1/k_1$, $1/k_2$ and $1/k_3$ that were used in the solution of the differential equation for both cases are found in Table II. Also in this table are the corresponding values of the coefficients k_1, k_2 and k_3. These coefficients, when multiplied by the instantaneous values of $\dot{p}_e$, p_e and $\int_0^t p_e \, dt$ respectively, describe the forces applied to the valve stem at any instant of time by the derivative control, proportional

control and integral control features of the servomechanism.

No attempt was made to design and build a servomechanism for this control problem due to the time limitations. However, steps were taken to obtain materials and equipment toward this end. A control valve similar to that presently installed, as seen in Figure 1, and a servomotor, the characteristics of which are shown on Plate IX, have been procured. It is planned that the project will be carried to completion using these items and the results of this study.

Results of the Case I solution as obtained from the various Brush recorder tapes are as follows:

Figure 7(a)	Maximum required control force:	13 lbs.
	Required steady state control force:	5.3 lbs.
(b)	p_e initial:	35.7 psi.
	p_e at $t = 1$ second:	3.0 psi.
	p_e at $t = 1.5$ seconds:	0.0 psi.
	p_e at $t = 2$ seconds:	1.2 psi.
	p_e at $t = 2.2$ seconds	0.0 psi.
Figure 8(a)	Coulomb friction varies from + 1.23 lbs. to -1.23 lbs. depending on the direction of motion of the valve. However, at about $t = 0.7$ second, amplifier 7	

apparently failed to saturate, so this term is in error past that time. It will be shown later that the coulomb friction effect is relatively small when compared to the other effects.

(b) This is the same trace as Figure 7(b), and was included for time comparisons.

Figure 9(a) Maximum required control force: 8 lbs.

Required steady state control force: 0.5lbs.

This plot and Figure 9(b) were included only as a matter of interest. They represent the condition in which the K_4F_d term is positive. This would correspond to a lift force on the control valve in the positive or down direction which was opposite to the lift force as measured.

(b) p_e initial: 35.7 psi.

p_e at t=1 second: 2.4 psi.

p_e at t = 1.5 seconds: 0.9 psi.

Figure 10

(a) $(\dot{p}_e/A_2)_{max} = (\dot{x})_{max}$: 1.1 ft/sec = 13.2 inch/sec.

$(\dot{p}_e)_{max}$ at t = 0.12 second: 653 psi/sec.

$\dot{p}_e$ at t = 1 second: negligible

(b)	This is the same trace as Figure 7(b), and was included for time comparison.
Figure 11(a)	$(\ddot{p}_e/n_2)$ max $= (\ddot{x})$ max: 22 ft/sec^2. Refer to equation (5) for the above relationship. The maximum acceleration occurs at zero time. It will be noted that the effect of the coulomb friction decreases the acceleration slightly when the velocity changes direction.
(b)	This is also the same trace as Figure 7(b), and was included for time comparison.
Figure 12(a)	This is the same trace as Figure 7(a), and was included for time comparison.
(b)	Maximum $\int_0^t p_e dt$ at t = 1.5 seconds: 8.3 lbs. $\int_0^t p_e dt$ steady state: 6.8 lbs. It is noted that in the steady state the only control force acting is that which is due to integral control. Comparing the steady state integral control force of 6.8lbs. with the steady state control force of 5.3 lbs. from Figure 7(a), a discrepancy of 1.5 lbs. is found. This disagreement is possibly due to the fact that $\dot{p}_e$ and p_e differ from zero by a small amount at the time selected for steady state comparison.

The Brush recorder tapes on which the solution to case II was recorded gave information as below. It should be remembered that ordinates that contain the factor n_2 in the dependent variable must be corrected for the variation of n_2 with time. Plate VIII, which is derived from Plate VI, is an aid in determining the value of n_2 at any time t.

Figure 13(a) Maximum required control force: 44 lbs.

Required steady state control force: 13 lbs.

(b) p_e initial: 21.5 psi.

$p_e$max at t = 0.75 second: 64.5 psi.

p_e at t = 1 second: 3.4 psi.

p_e at t = 1.2 seconds: negligible

Figure 14(a) The coulomb friction force varies in a manner similar to that of Figure 8(a). Amplifier 7 apparently failed to saturate at about t = 0.6 second in this case.

(b) This trace is identical to Figure 13(b), and was included for time comparison.

Figure 15(a) $(\dot{p}_e/n_2)$max = $(\dot{x})$max: 1.3 ft/sec = 15.6 in./sec.

$(\dot{p}_e)$max at t = 0.09 second: 770 psi/sec.

$\dot{p}_e$ at t = 0.6 second: negligible.

(b) This trace is identical to Figure 13(b), and was included for time comparison.

Figure 16(a) $(\ddot{p}_e/n_2)$max = $(\ddot{x})$max at t = 0.0 second: 43 ft/sec^2

The small effect of coulomb friction is noted in this trace also.

(b) This trace is identical to Figure 13(b), and was included for time comparison.

Figure 17(b) Maximum $\int_0^t p_c\,dt$ is at $t = 1.2$ seconds: 3.15 lbs.

ELECTRONIC DIFFERENTIAL ANALYZER DISCUSSION

Complicated circuits should not be attempted by those unfamiliar with computer operation. It is recommended that a simplified version of the problem be solved first, so that a degree of familiarity with the complexities of the computer might be obtained by the operator before proceeding to more difficult circuits.

The usual mathematical methods for solving second order differential equations such as are contained in this report would prove to be extremely laborious and time consuming. On the other hand, the computer seems to be an ideal means for obtaining such solutions. The most difficult part of the procedure is the setting up of the equipment to duplicate the equation. Then the values of the constants are obtained by a trial and error process, noting the response of the oscillograph. When a satisfactory response has been obtained, the constants which caused this response are known, and the problem is solved. The constants thus determined can be used to design the actual control system.

Resistors and shorting bars were mounted on General Radio type 274 - M double plugs which fitted into the terminals of the amplifiers. Erratic results from time to time were finally traced to the looseness of the mechanical connections between the resistors and plugs. Time can be saved by checking these connections before inserting the plugs into the amplifier terminals.

The amplifiers used are of an inexpensive type constructed several years ago and must be individually balanced for zero output. Allowing the amplifier tubes to heat up for a long period of time has a favorable effect on maintaining balance. It is recommended that the tubes be allowed to operate overnight if possible. Room temperature and humidity also effect balance, and where their variation is not closely controlled, frequent balance checking is necessary. Of course, a well regulated power supply for the amplifiers is also necessary for good balance.

CONCLUSIONS

The objective of this study was to determine the characteristics of such a control system as would reduce the pressure error in the settling chamber of a particular blowdown wind tunnel to within plus or minus one pound per square inch within one second. By referring to Figure 13 it may be seen that the solution obtained in this study can be considered as in satisfactory agreement with the above stated objective. The values of the constants associated with the solution are found in Table II.

It is recommended that the design and fabrication of a servomechanism that will exhibit the characteristics as set forth in Table II be accomplished, so that satisfactory pressure control in the subject wind tunnel may be obtained.

REFERENCES

Erickson, W.A. and
Parrett, G.S.

The Use of an Electronic Analog Computer in the Determination of the Normal Modes of Lateral Vibration of Non-uniform Beams.
A thesis submitted to the Aeronautical Engineering Dept. of the University of Michigan, June, 1949.

Frost, S.

Compact Analog Computer
Electronics Magazine, July 1948, pp 116-122.

Hagelbarger, D.W.,
Howe, C.E. and
Howe, R.M.

Investigation of the Utility of an Electronic Analog Computer in Engineering Problems
The Engineering Research Institute, Ann Arbor, Michigan 1949, 1st ed.

Korn, G.A.

Elements of D-C Analog Computers
Electronics Magazine, April 1948, pp 122-127

Ragazzini, J.R.,
Randall, R.H. and
Russell, F.A.

Analysis of Problems in Dynamics by Electronic Circuits
Proceedings of the IRE, 1947 Vol. 35, pp 444 - 452, May, 1947.

COMPUTATIONS

DRIVE COMPUTATIONS

Data:

Weight of valve and stem:	10.38 lbs.
Coulomb friction force:	1.23 lbs.
"Sticktion" force:	approx. 1.00 lbs.
Valve aerodynamic lift	See Plate II.

Assumed configuration: See Fig. 2.

Acceleration force: Assuming a maximum acceleration of 6 in/sec^2,

$$\text{Force} = m \times a = \frac{(10.38)}{(32.2)}(0.5) = 0.254 \text{ lbs.}$$

Total Force: Neglecting Lift Force other than the 6 lbs. for which the valve is compensated,

$$F_T = F_A + F_C + F_S = 0.25 + 1.23 + 1.00 = 2.48 \text{ lbs.}$$

Work/sec required of motor: $\frac{2.48(0.75)}{(12)(.5)} = 0.31 \frac{\text{ft. lbs.}}{\text{sec.}}$

Required Horsepower: 0.000564 HP.

Motor Data: Diehl, Navy type CDA [illegible]11012, Max. output = 5.2 watts or 0.00698 HP @ 3.5 oz. in. torque and 2000 rpm.

If motor is to accelerate between zero and 2000 rpm in ½ second, the torque would vary from 5.46 to 3.5 oz. inches. This would be equivalent to operating at 4.7 oz. inches and 1000 rpm constantly for one half second.

$$\text{Work/sec} = \frac{(4.7)(1000)(2 \times 3.14)}{(12)(16)(60)} = 2.56 \frac{\text{ft. lbs.}}{\text{sec}}$$

This compares to 0.31 ft. lbs./sec required by the 2.48 lb. load.

Thus the motor would deliver a force of 20.5 lbs. at the valve stem. $\frac{x}{2.56} = \frac{2.48}{0.31}$, $x = 20.5$ lbs.

In order to exert a 20.5 lb. force on the valve stem, the average torque of 4.7 oz. inches must act through a gear train: $\frac{(4.7)}{(16)} x = 20.5$, $x = 69.7$ gear ratio.

After a solution to CaseI was found, greater control forces were required to give a larger margin of available control motor force. The Diehl, Navy type CDA 211426 servomotor was available, which would operate between 8.4 and 11.5 oz. inches over the same rpm range. The equivalent constant torque in this case is 10.4 oz. inches, and the motor would deliver an average force of 56.7 pounds to the valve stem. The gear ratio necessary for this motor would be 87 to 1. $\frac{(56.7)(16)}{(10.4)} = 87.0$ gear ratio.

Principle Dimensions of
University of Michigan Blow Down
Wind Tunnel

Volume of storage tank:	100	ft.^{3}
Maximum supply line pressure:	95	psig.
Nozzle throat section area:	0.28	in.^{2}
Nozzle test section area:	1.00	in.^{2}

All plumbing between storage tank and nozzle is 3 inch pipe and fixtures.

TABLE I

Control Force Constants

	$1/k_1$ ohms	$1/k_2$ ohms	$1/k_3$ ohms	k_1 $\frac{\# \text{ sec}}{\text{ft}}$	k_2 $\frac{\#}{\text{ft}}$	k_3 $\frac{\#}{\text{ft sec}}$	K_1 $\frac{\# \text{ sec}}{\text{psi}}$	K_2 $\frac{\#}{\text{psi}}$	K_3 $\frac{\#}{\text{psi sec}}$
Case I	0.3	0.008	0.003	3.33	125	333	0.0056	0.21	0.56
Case II	0.15	0.003	0.001	6.67	333	1000	0.0112	0.56	1.68

TABLE II

Settling Chamber Pressure vs. Valve Position

x inches	0.125	0.375	0.625	0.875	1.125	1.375
p_{t_i} 93 psig	41.0	53.5	66.0	76.0	79.0	80.0
p_{t_i} 88 psig	36.8	47.8	59.2	69.4	72.3	73.2
p_{t_i} 76 psig	33.3	43.0	54.5	60.0	63.3	63.5
p_{t_i} 60 psig	24.8	32.5	42.5	48.8	49.5	51.0

p = psig

TABLE III

Lift Force vs Valve Position

x in.	Lift lbs.
0.5625	10.19
0.6875	11.59
0.750	13.50
0.875	14.50
1.000	17.00
1.125	17.175
1.375	20.00

TABLE IV

Determination of Coulomb Friction Force and Weight

Up Motion lbs.	Down Motion lbs.
11.500	9.000
11.687	9.313
11.844	9.063
11.406	9.313
11.656	9.063
11.625	9.188
11.563	9.156
Avg: 11.61[illegible]=W+F	Avg: 9.156=W-F

W = 10.38 lbs.

F = 1.23 lbs.

TABLE V

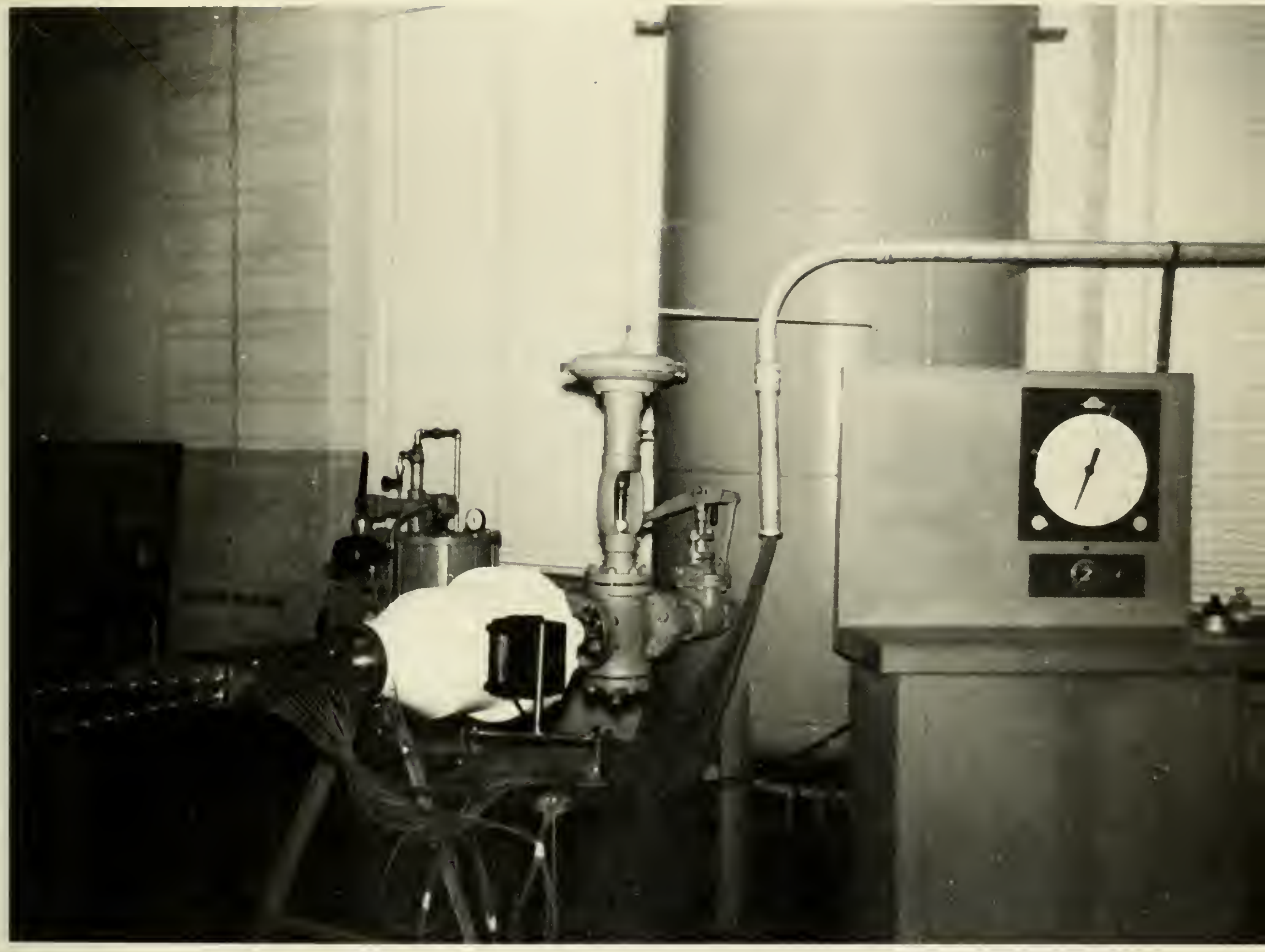
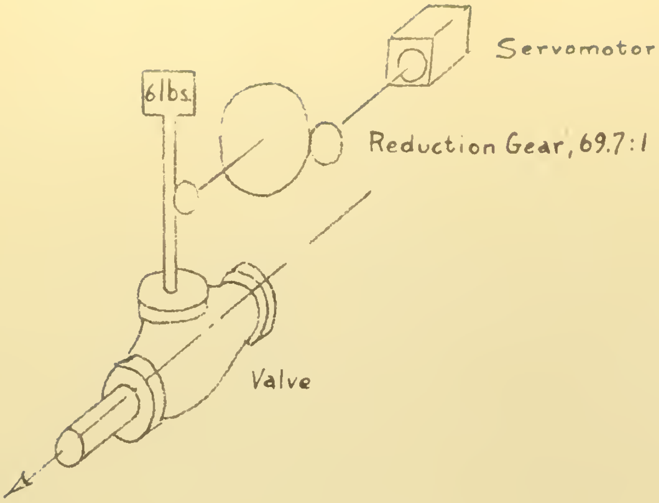
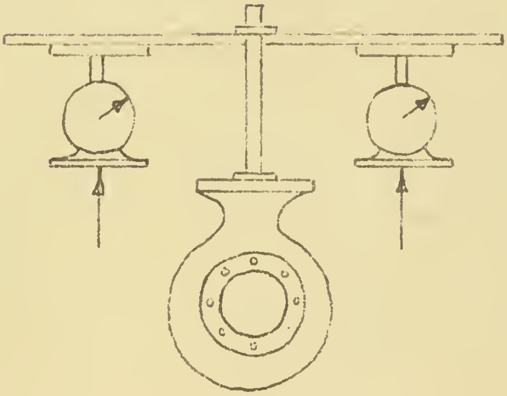


Fig. 1



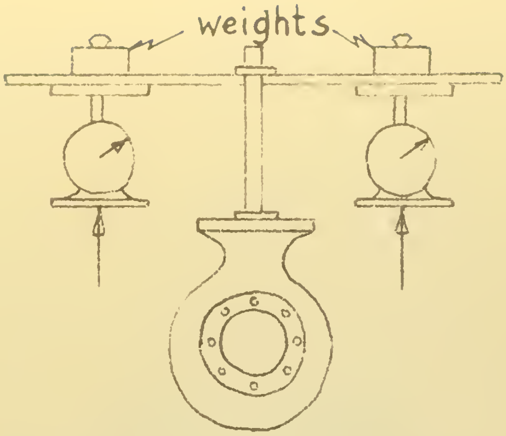
Sketch of Proposed Drive Mechanism

FIG. 2



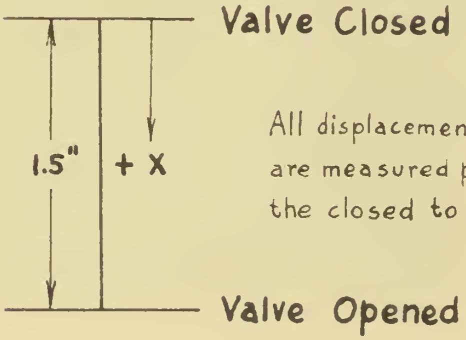
Sketch Indicating Method Used to Measure Coulomb Friction and Weight of Valve

FIG. 3



Sketch Indicating Method Used to Measure "Lift" Force of Valve

Fig. 4



All displacements and forces are measured positive from the closed to the opened position.

Coordinate System

Fig. 5

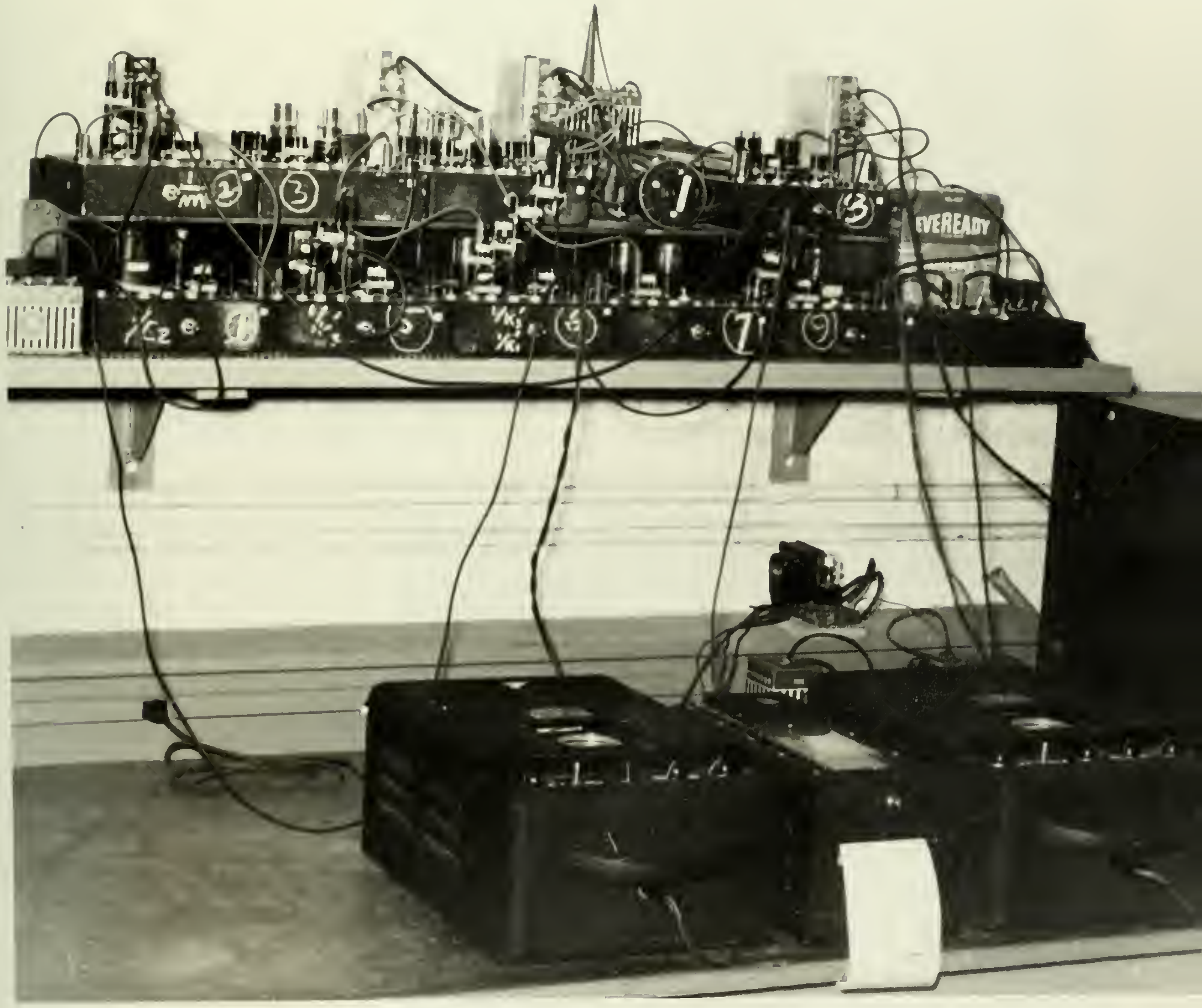


Fig. 6

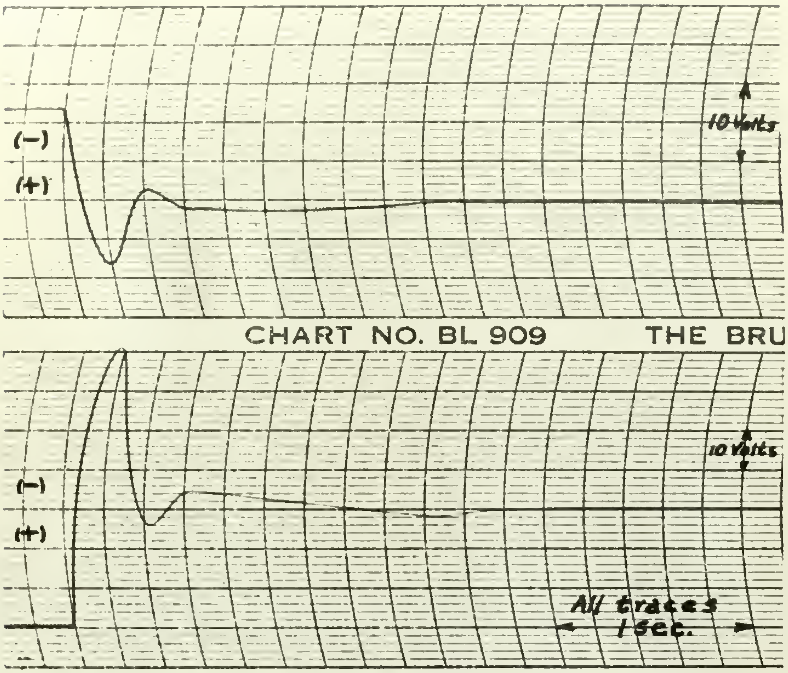


Fig. 7(a): Control Force. $1/k_1 = 0.3$; $1/k_2 = 0.008$; $1/k_3 = 0.008$, all in megohms.

Fig. 7(b): $\frac{p_e}{n_2} \times 500$. $K_4 Pd = -6$ volts. Constants same as 7(a).

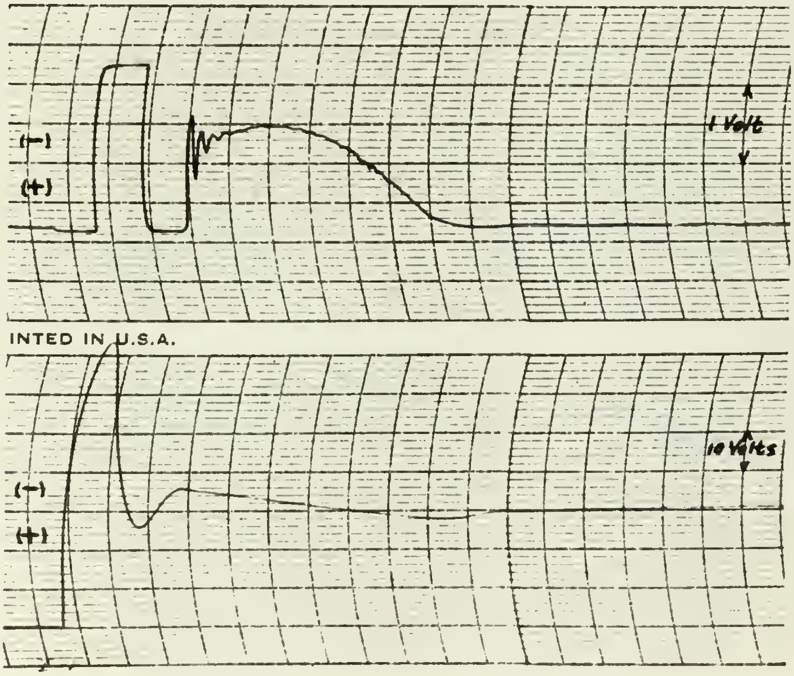
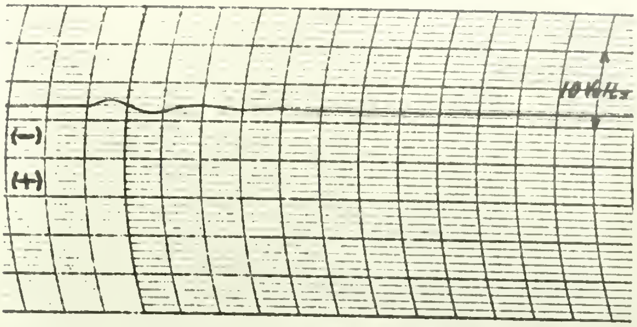


Fig. 8(a): Coulomb Friction Force. Constants same as Fig. 7(a).

Fig. 8(b): Same as Fig. 7(b).



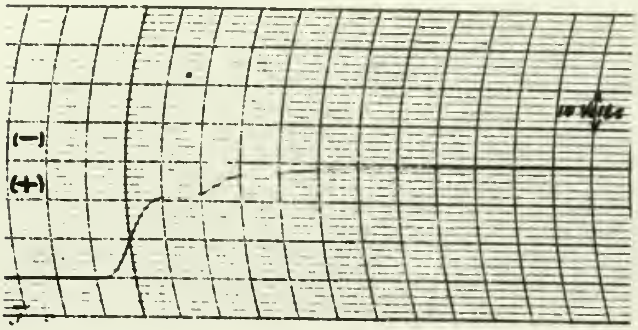
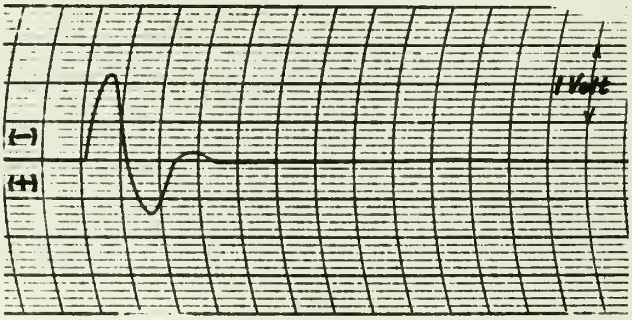


Fig. 9(a): Control Force. Constants same as Fig. 7(a).

Fig. 9(b): $\frac{P_e}{n_2}$ x 500. $K_4Pd = +6$ volts. Constants same as 7(a).



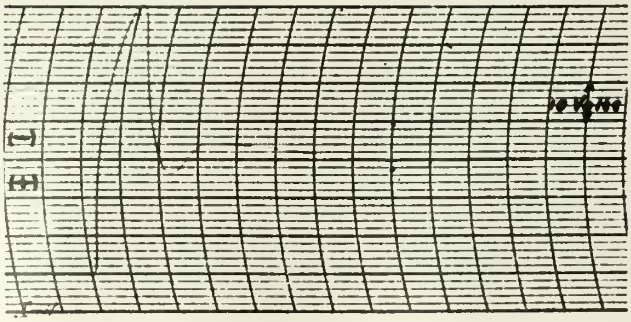


Fig. 10(a): $\frac{\dot{P}_e}{n_2}$. Constants same as in Fig. 7(a).

Fig. 10(b): Same as Fig. 7(b).

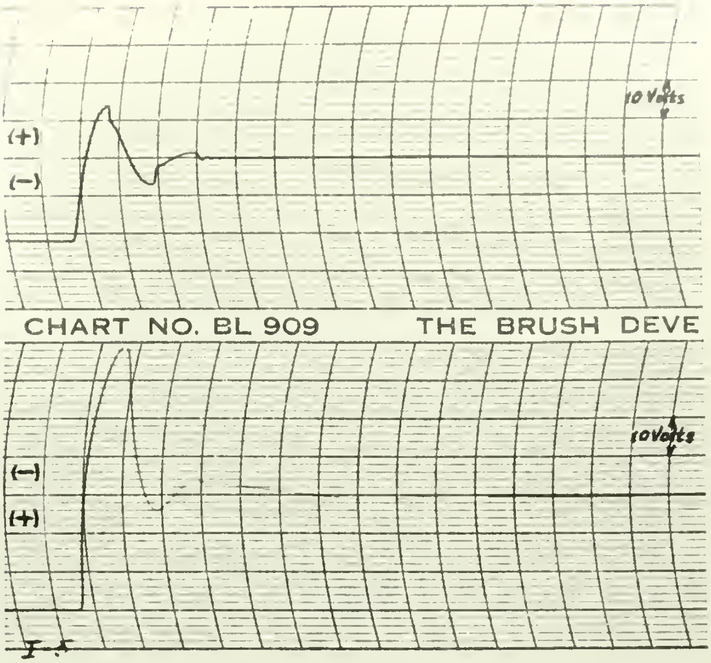


Fig. 11(a): $\frac{\ddot{p}_e}{n_2}$. Constants same as in Fig. 7(a).

Fig. 11(b): Same as Fig. 7(b).

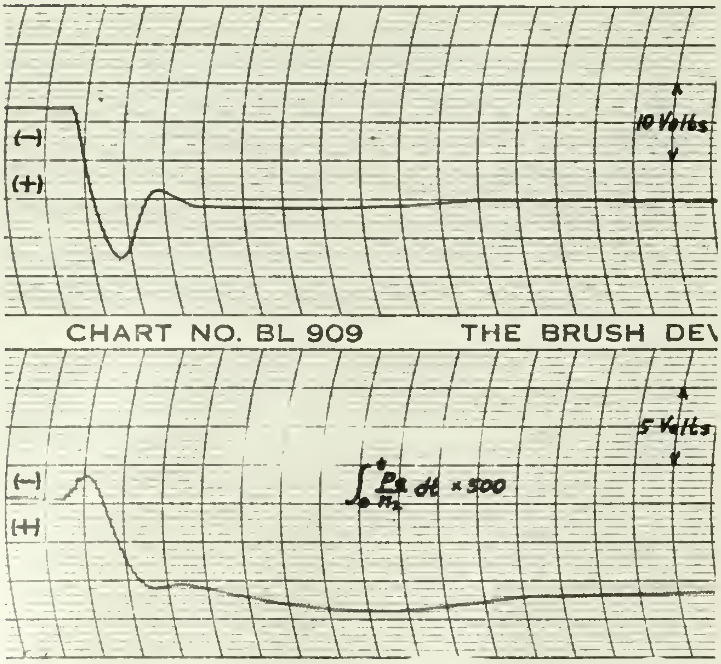
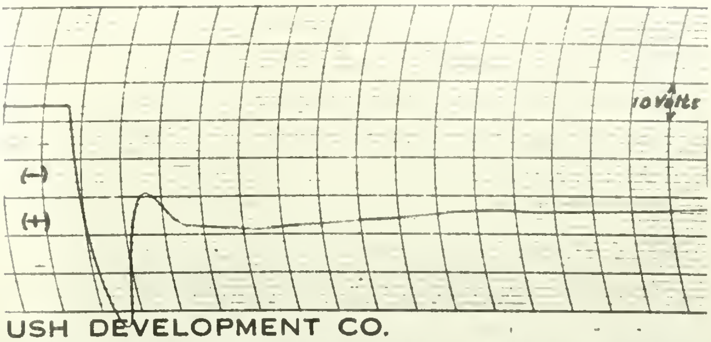


Fig. 12(a): Same as Fig. 7(a).

Fig. 12(b): Integral Force x 500. Constants same as 7(a).



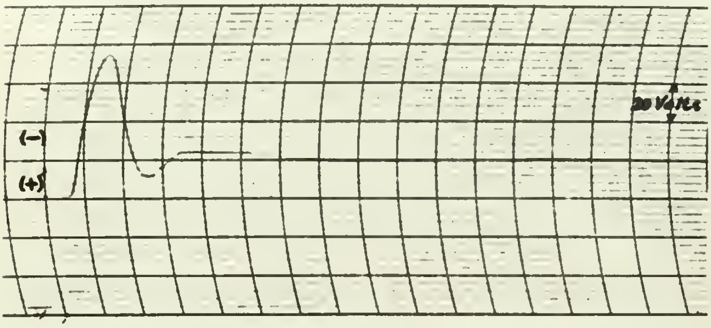
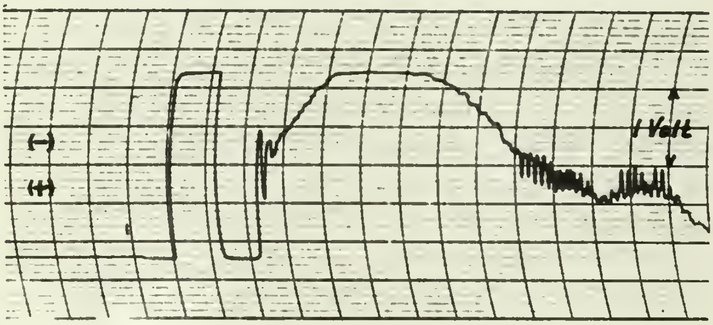


Fig. 13(a): Control Force. $1/k_1$ 0.15; $1/k_2$ 0.003; $1/k_3$ 0.001, all in megohms.

Fig. 13(b): $\frac{p_e}{n_2}$ x 500. Constants same as Fig. 13(a).



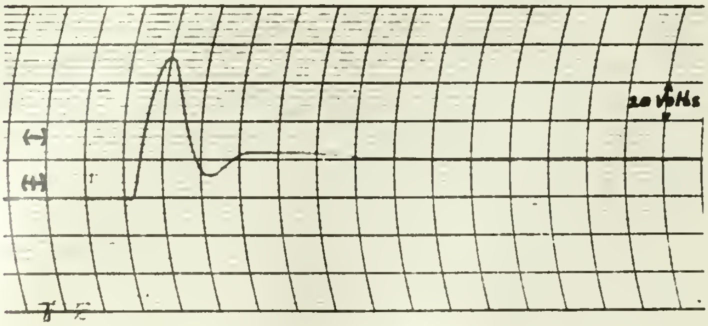


Fig. 14(a): Coulomb Friction Force. Constants same as Fig. 13(a).

Fig. 14(b): Same as Fig. 13(b).

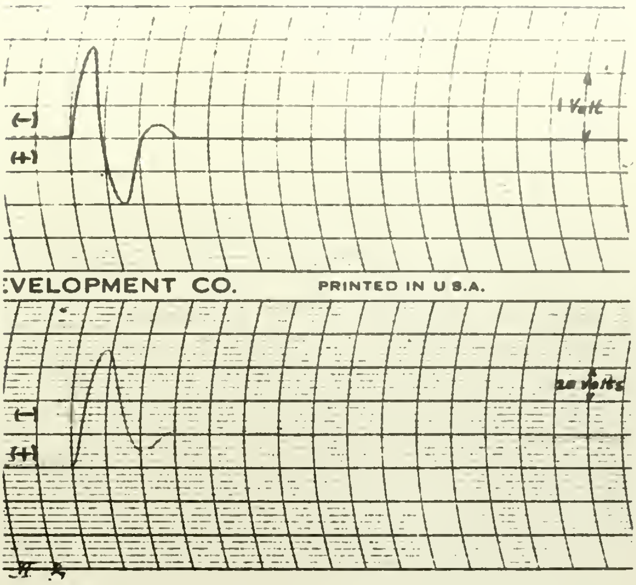


Fig. 15(a): $\frac{\dot{p}_e}{n_2}$. Constants same as 13(a). (b): Same as 13(b).

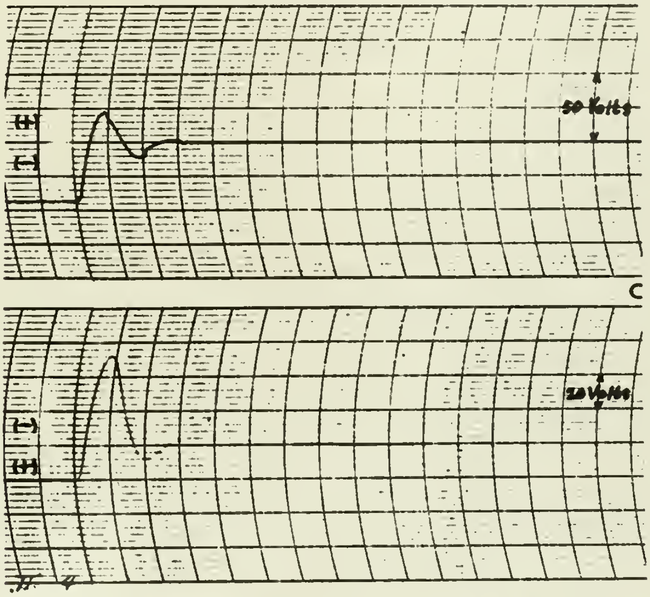


Fig. 16(a): $\frac{\ddot{p}_e}{n_2}$. Constants same as 13(a). (b): Same as 13(b).

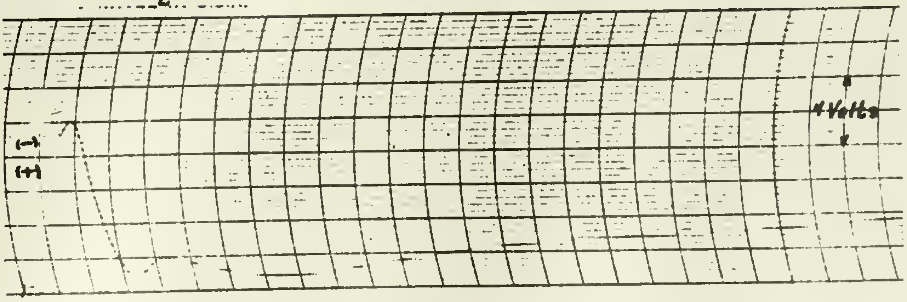


Fig. 17: $\int_0^t \frac{p_e}{n_2}\,dt \times 500$. Constants same as Fig. 13(a).

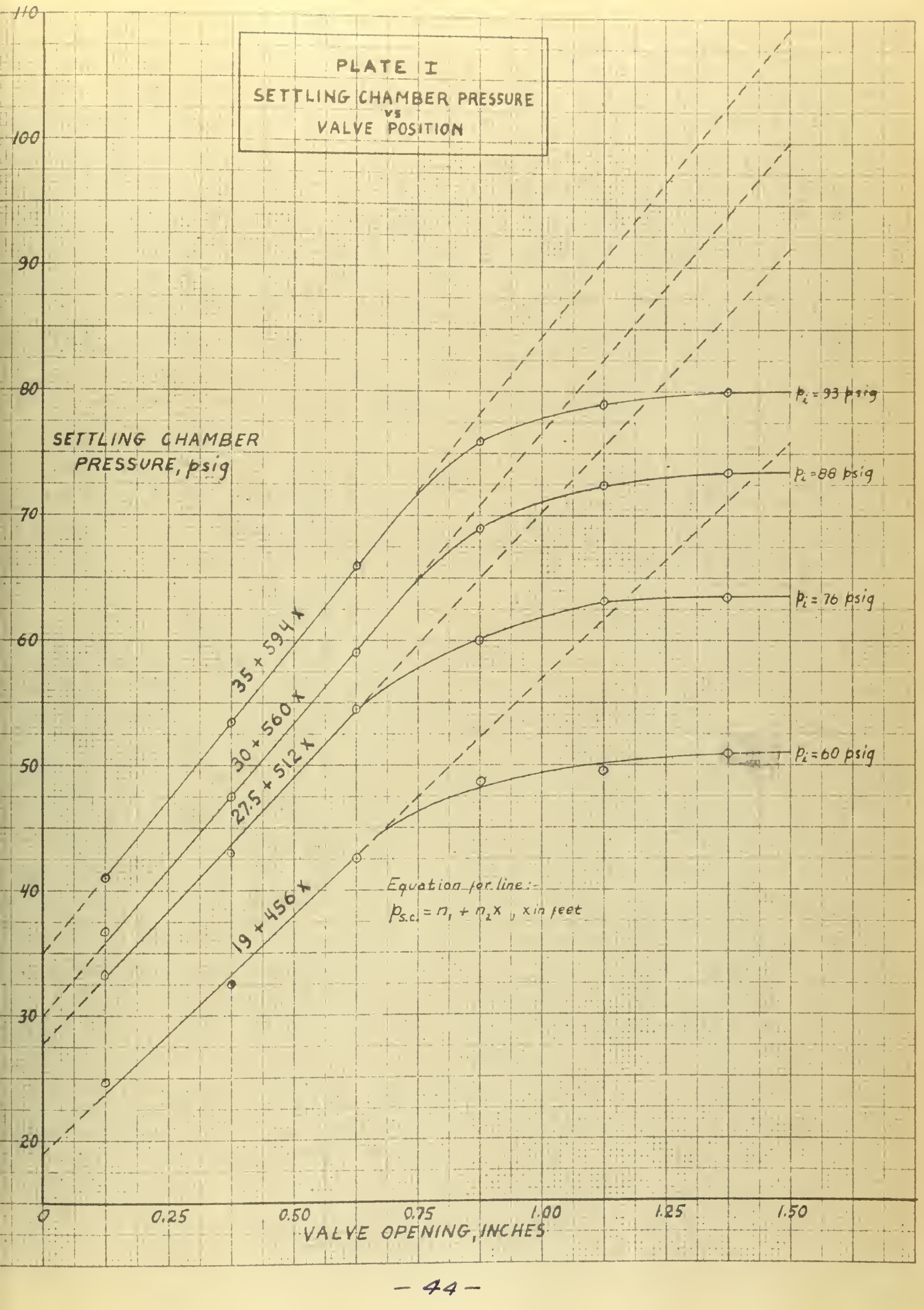


PLATE I

SETTLING CHAMBER PRESSURE vs VALVE POSITION

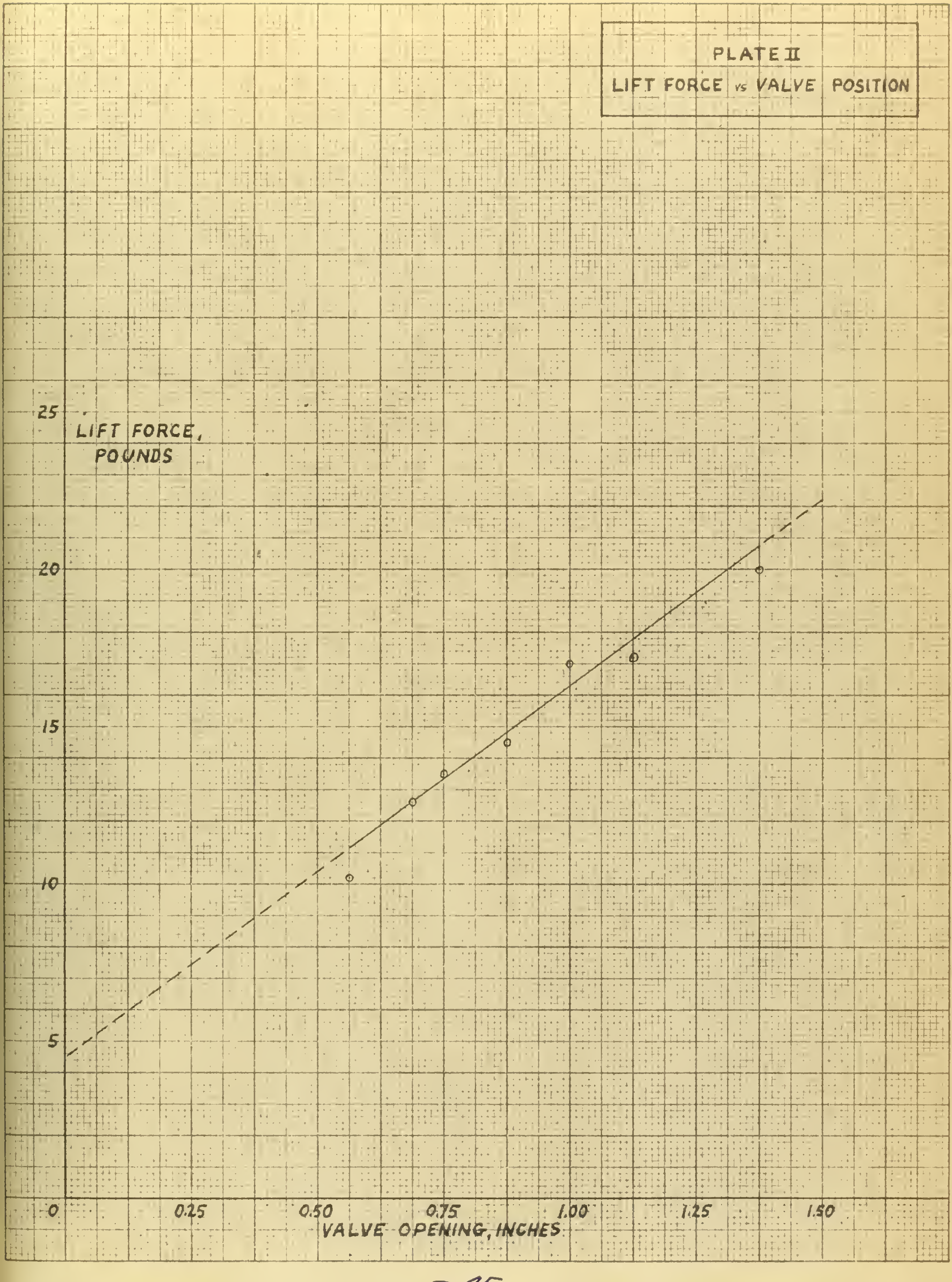


PLATE II

LIFT FORCE vs VALVE POSITION

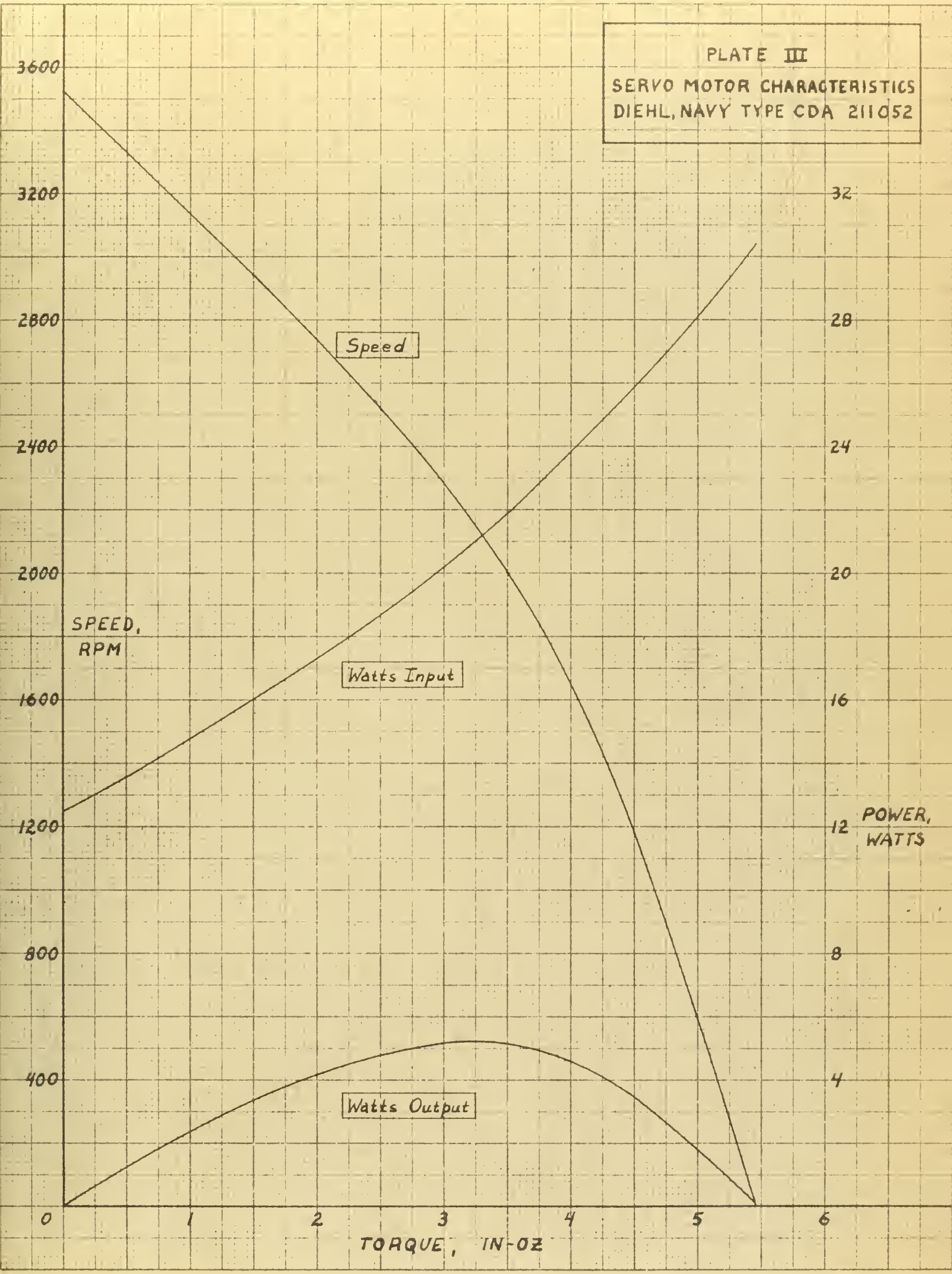
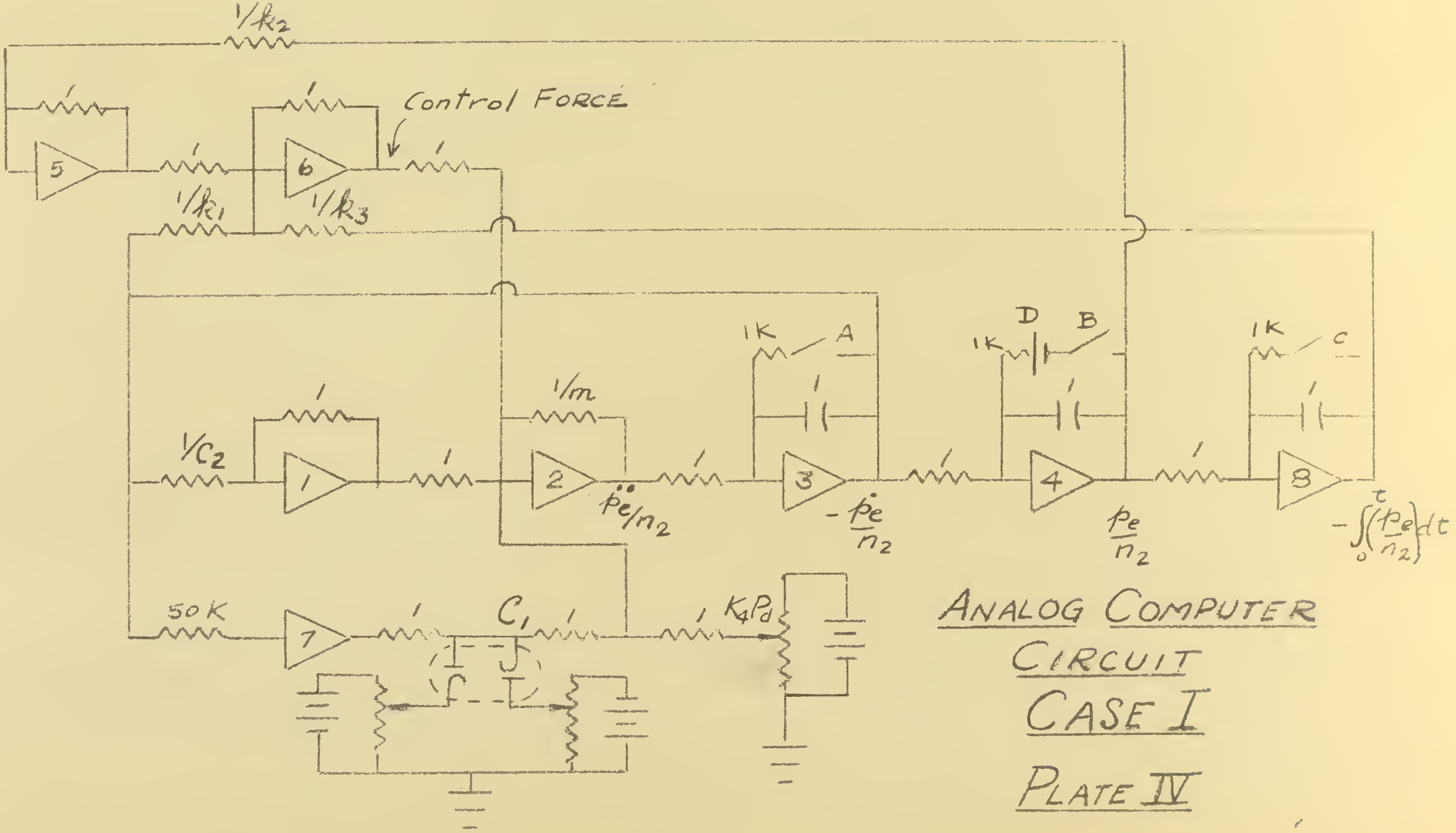


PLATE III
SERVO MOTOR CHARACTERISTICS
DIEHL, NAVY TYPE CDA 211052
Speed
Watts Input
Watts Output
SPEED, RPM
POWER, WATTS
TORQUE, IN-OZ
3600
3200
2800
2400
2000
1600
1200
800
400
0
32
28
24
20
16
12
8
4
1
2
3
4
5
6

$$m\left(\ddot{p}_e/n_2\right) + C_1 \frac{\dot{p}_e}{|\dot{p}_e|} + C_2\left(\dot{p}_e/n_2\right) + k_1\left(\dot{p}_e/n_2\right) + k_2\left(p_e/n_2\right) + k_3\int_0^t\left(p_e/n_2\right)dt + K_4 P_d = 0$$



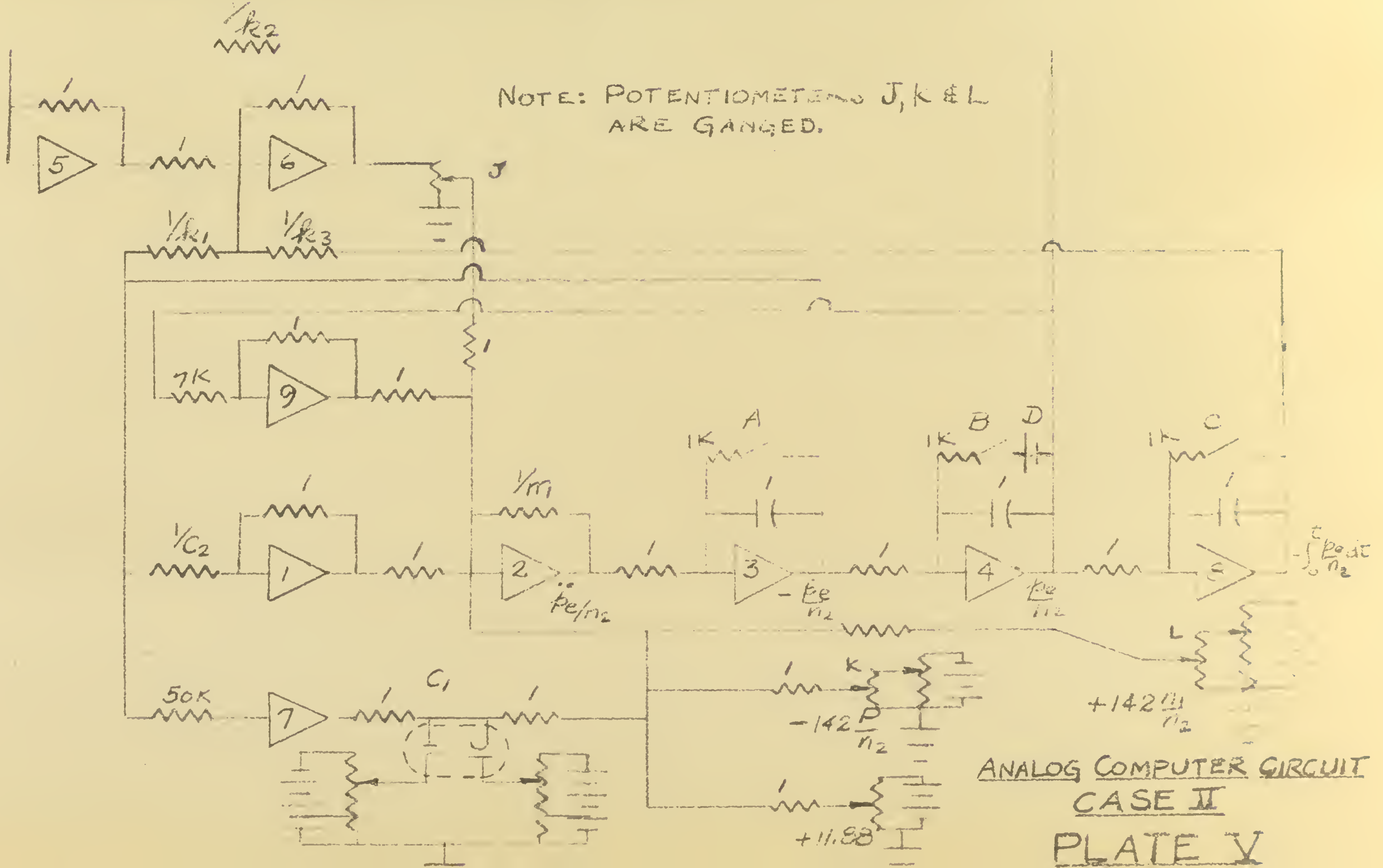
ANALOG COMPUTER

CIRCUIT

CASE I

PLATE IV

$$m\left(\frac{\ddot{p}_e}{n_2}\right) + C_1\frac{\dot{p}_e}{|\dot{p}_e|} + C_2\left(\frac{\dot{p}_e}{n_2}\right) + k_1\left(\frac{\dot{p}_e}{n_2}\right) + k_2\left(\frac{p_e}{n_2}\right) + k_3\int_0^t\left(\frac{p_e}{n_2}\right)dt + 11.88 - 142\frac{p_e}{n_2} - 142\frac{P}{n_2} + 142\frac{n_1}{n_2} = 0$$



NOTE: POTENTIOMETERS J, K & L
ARE GANGED.

ANALOG COMPUTER CIRCUIT
CASE II
PLATE V

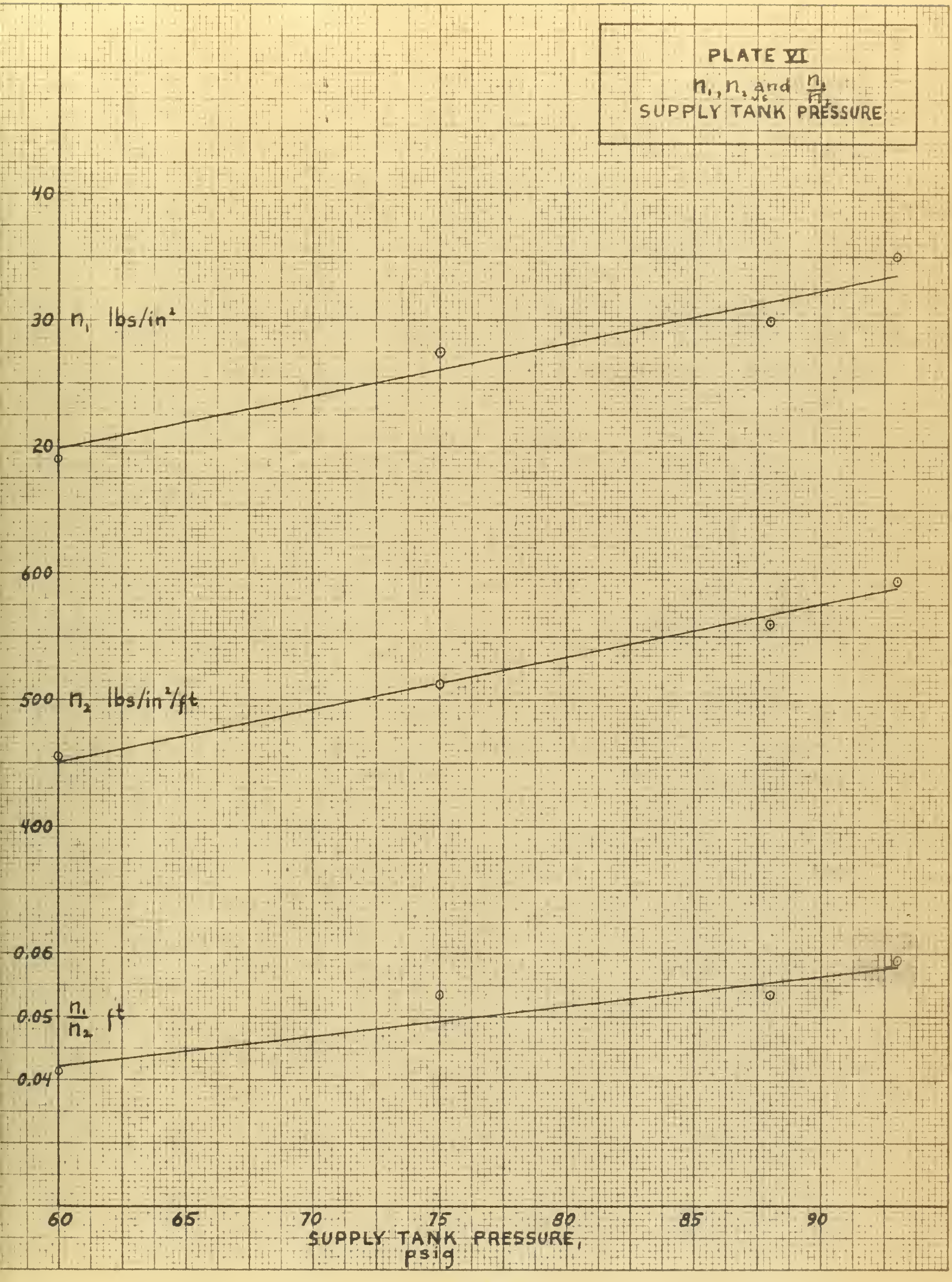


PLATE VI

n_1, n_2 and $\frac{n_1}{n_2}$ vs SUPPLY TANK PRESSURE

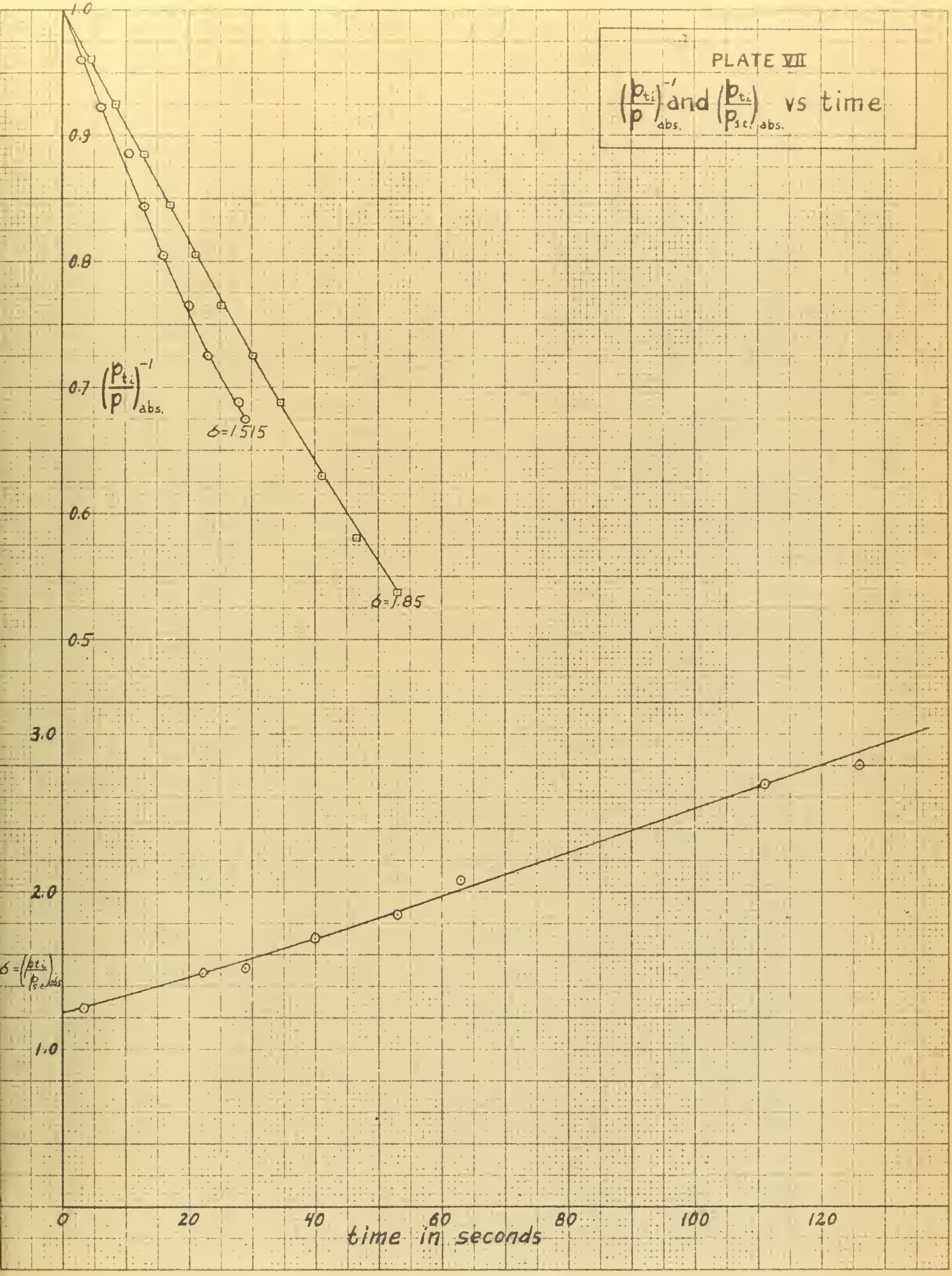


PLATE VII

$\left(\frac{p_{ti}}{p}\right)^{-1}_{abs.}$ and $\left(\frac{p_{ti}}{p_{sc}}\right)_{abs.}$ vs time

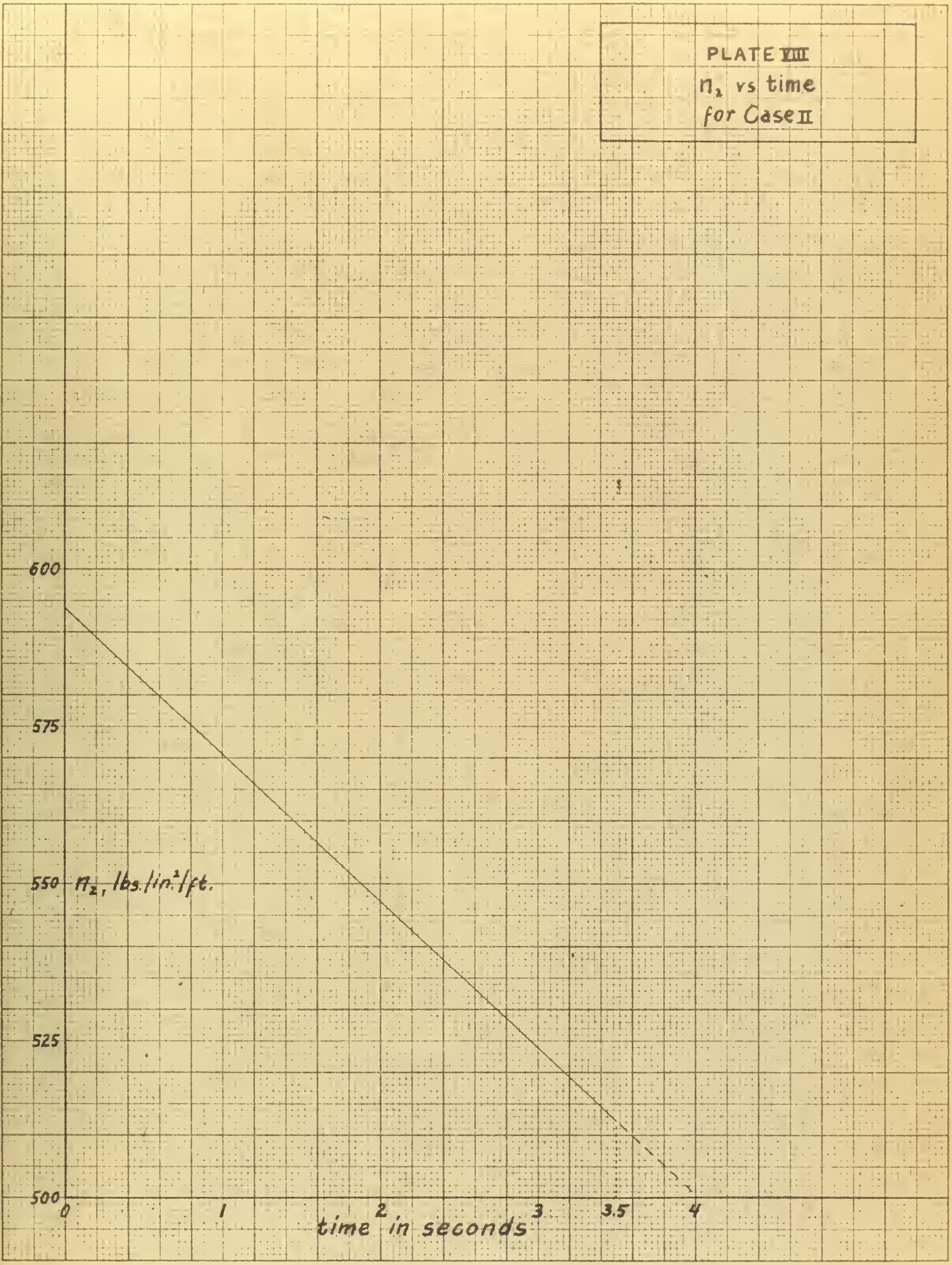


PLATE VIII
Π_2 vs time
for Case II
600
575
550
Π_2, lbs./in.²/ft.
525
500
0
1
2
3
3.5
4
time in seconds

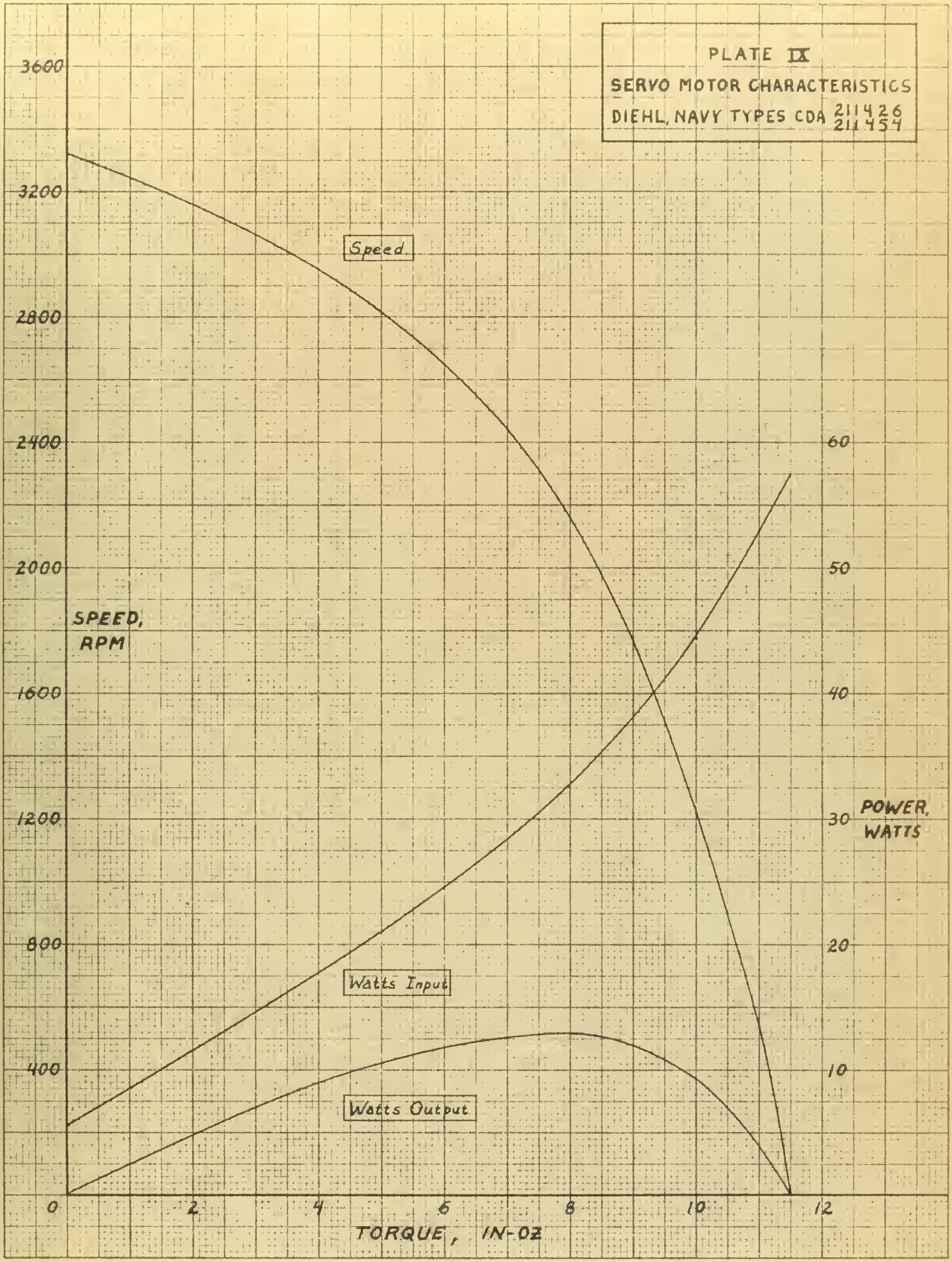


PLATE IX

SERVO MOTOR CHARACTERISTICS

DIEHL, NAVY TYPES CDA 211426
211454

DATE DUE

APR 20 510

12984
Thesis
K13 Kanaga
AUTHOR
Study of pressure control for a
blowdown wind tunnel
BORROWER'S NAME

USE ACCO PAPER FASTENERS FOR BINDING
ACCOPRESS BINDER
No. BH 2507
MADE BY
ACCO PRODUCTS INCORPORATED
OGDENSBURG, N.Y. U.S.A.

thesK13
A study of pressure control for a blowdo
3 2768 002 11391 2
DUDLEY KNOX LIBRARY